



On the integration between fluidized bed and Stirling engine for micro-generation



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HIGHLIGHTS

- High heat-exchange rate in fluidised bed is appealing for coupling a Stirling engine.
- A mathematical model of a coupled system newly developed.
- The “integrated system” is of interest for micro-scale co-generation (1 kW) from renewable fuels.
- A PID control strategy largely enhances the system performances.

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ABSTRACT

The present article deals with the integration of a fluidized bed combustor and a Stirling engine for cogeneration purposes. An experimental study was carried out, proving the ability of the bed to exchange heat at high rate with an immersed coil that realistically emulates the heat exchanger of a small Stirling engine. The heat transfer coefficient attains values up to $280 \text{ W m}^{-2} \text{ K}^{-1}$. No dirtying of the immersed surface occurred during a combustion test of biomass.

The paper also reports on a newly developed mathematical model of a fluidized bed combustor coupled with a Stirling engine for co-generation purposes. It consists of four fundamental blocks describing i) the heat transfer, ii) the fluidized bed combustion, iii) the heat recovery, and iv) the Stirling engine. The model produces as relevant outputs the bed temperature, the mechanical power and the efficiency of the Stirling engine, at changing the operating conditions and geometrical parameters of the system. A slow dynamic response is predicted, that it is significantly improved by adopting an efficient control strategy. On the whole, the model results indicate that the proposed “integrated system” is of interest for micro-scale cogeneration from biomass fuels.

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1. Introduction

The biomass fuels have intrinsic CO_2 neutral impact, independently of the way their chemical energy is transformed into heat or power via thermal or biological processes. In contrast, techno-economic constraints limit the viability of biomass utilization for power generation.

From a technical point of view, although the combustion of biomass fuels is relatively easy to be carried out in both fixed and fluidized bed devices, the process implementation at small scale is not straightforward and efficient for power generation. For instance, the minimal size of organic Rankine cycle (ORC) is

normally in the range 100–200 kW_{el} [1] and the thermal efficiency is lower than 20% [2]. Furthermore, capital costs are high at small scale applications. The cogeneration can be also accomplished by coupling a gasification/pyrolysis step with an internal combustion engine or fuel cell, but this option results more complex due to the required purification steps of the syngas [3]. Recently, an increasing attention is paid to the application of micro-scale cogeneration from different fuels, thanks to the diffusion of even more dense grids for electricity distribution and picking up [4]. In this scenario, the development of reliable and cheap methods for cogeneration from biomass fuels in a range 1–100 kW_{el} is appealing, representing a still open challenge.

The external combustion Stirling engine [5] has been conceived and developed since the 19th century, although having never reached a high level of penetration into the energy and transport sector. Main advantages are the smoothness, the flexibility toward the external heat source and the high thermodynamic efficiency.

Abbreviations: PID, proportional-integrative-derivative control; FB, fluidized bed; SE, Stirling engine.

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The Stirling engine can easily make possible micro-scale cogeneration from clean fuels (e.g. natural gas) for both stand-alone and on-grid applications. In contrast, dirtying/fouling problems arising from deposition of unburned species, soot and ash over the surface of the heat exchanger are likely to occur, in particular when low grade fuels are burnt [6]. As consequence, the system reliability could be affected and the performances worsened. A typical application of Stirling engine was recently proposed at small scale by Li et al. [7] for power generation by waste gases from an IC engine. Nevertheless, this option should be proven to be reliable upon long operation time because the soot deposition could seriously affect the performance of the complex heat exchanger they adopted.

The fluidized bed (FB) combustion boasts many advantages, as the flexibility toward the fuel characteristics and excellent thermal properties [8]. In particular, fluidized beds systems were proved to be effective and reliable for burning biomass and agriculture residues [9]. Albeit more complex than other options, a fluidized bed combustor easily achieves limited CO emissions and high carbon conversion even at small scales [10]. When biomass having high alkali content is used, fouling of cold parts or agglomeration of the bed particles can be experienced, demanding the adoption of proper measures (e.g. alternative bed materials, improved internal mixing) [11]. Furthermore, the fluidized bed exerts a self-cleaning action of all surfaces that are in contact with the moving bed particles preventing dirtying/fouling problems [12].

Coupling a fluidized bed combustor with a Stirling engine beneficiaries of very high heat transfer coefficients and reduced surface fouling in the bed region [13]. Even in presence of complex tube arrays, the abrasion exerted by the bed material contributes to clean the external surface of the tubes and to remove char/ash particles locked in interstices. Currently, there are no published experiences and results on direct coupling of a Stirling engine with a fluidised bed combustor.

The present article provides insights into the integration of a fluidized bed combustor and a Stirling engine for cogeneration. An experimental approach was adopted for checking the ability of the bed to exchange heat with an immersed coil, simulating in realistic way the heat exchanger of a Stirling engine. In parallel, a mathematical model of a FB combustor coupled with a Stirling engine was developed. It computes the mechanical power and the efficiency of the engine at changing the operating conditions and geometrical parameters of the whole system. The dynamic response, as consequence of start-up or changes in the working regime, is also predicted. The results of the research activity are reported and discussed in the paper with the aim to support a parallel setup of an experimental prototype.

2. Experimental assessment of the heat transfer

2.1. Equipment

The experimental apparatus for measuring the heat transfer coefficient is displayed in Fig. 1. The fluidization column is a 120 mm ID, 500 mm high tube, made by steel. It is heated by means of an external electric furnace having a total power of around 4 kW. The air distributor is formed by a fixed bed of 3/8" stainless steel spheres. The primary air stream, metered by means of an electronic mass-flow controller, passes through the fixed bed before fluidizing the bed. The bed temperature (T_b) is monitored by a submerged K-type thermocouple.

A coil is immersed in the bed for heat transfer measurements. The coil is formed by an AISI 304 tube, 4 mm ID and 6 mm OD. The tube is suspended from the open top of the equipment and is purposely bended as shown in the right of Fig. 1. Since the two

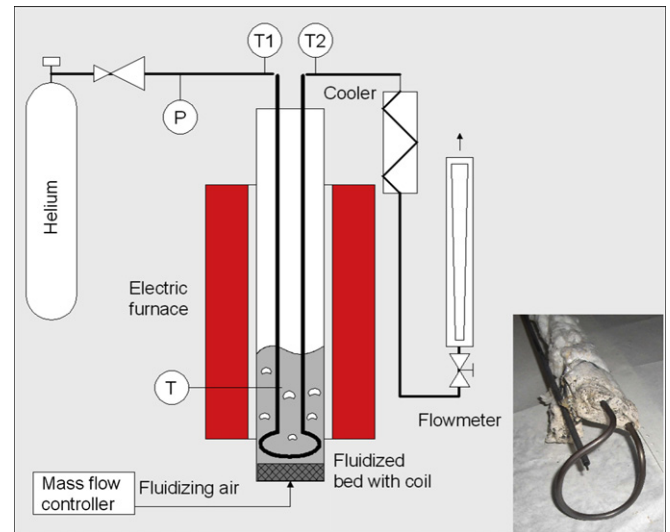


Fig. 1. Experimental apparatus for heat transfer experiments in fluidized bed.

vertical legs are thermally insulated, the length of the tube in direct contact with the hot bed is 300 mm only.

Silica sand of two different sizes (0.2–0.4 and 0.3–0.6 mm) was used as bed material under bubbling fluidization regime in air with a total expanded height of about 200 mm.

The gas entering the immersed tube is helium provided by a bottle, after a pressure reduction stage. A flow meter is used for measuring the gas flow rate (F_{He}) downstream a cooling stage and a valve for flow regulation. Two thermocouples measure the stream temperature immediately before (T_1) and after (T_2) the immersed coil.

The same equipment was used for verifying the occurrence of fouling on the tube surface by feeding a biomass fuel (olive husks) for 2 h.

2.2. Results

The results of the experiments are displayed in Figs. 2 and 3. The heat transfer coefficient h is calculated upon elaboration of the time averaged temperatures T_1 and T_2 via Eq. (1), where $c_{p,He}$ is the specific heat of the helium, D_t and L_t are the external diameter and the exposed length of the tube, respectively.

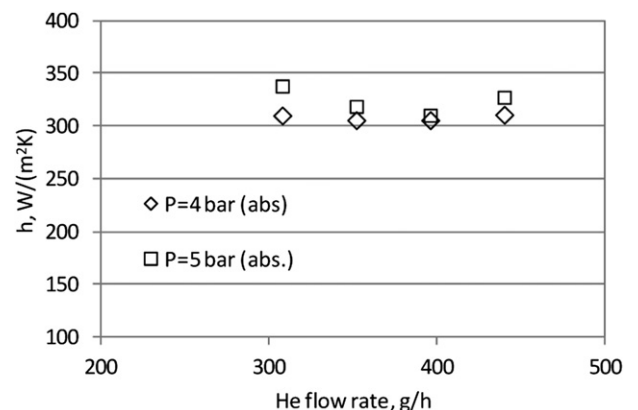


Fig. 2. Heat transfer coefficient versus the He flow rate at different pressure in the coil (fine sand, $U = 0.3$ m/s).

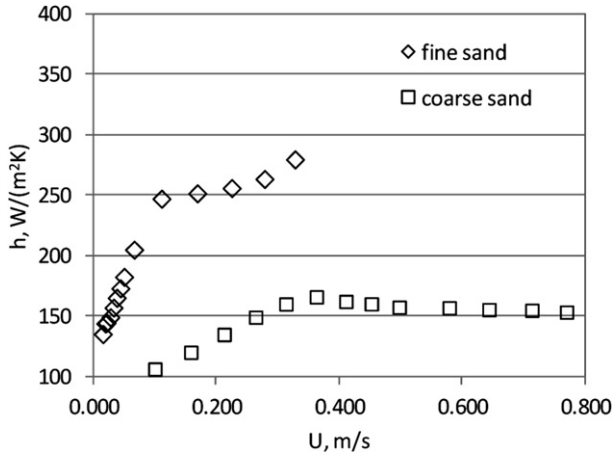


Fig. 3. Heat transfer coefficient versus the fluidization velocity for fine and coarse bed materials ($P = 4$ bar abs.).

$$h = \frac{c_{p,He} F_{He} (T_2 - T_1)}{\pi L_t D_t} \left(\frac{1}{T_b - \frac{T_2 + T_1}{2}} \right) \quad (1)$$

The coefficient h is substantially independent of both internal coil pressure and He flow rate (Fig. 2), indicating that the internal resistance to the heat transfer is smaller than the external one for the test conditions. The Re number in the coil for He flow rate = 440 g/h is equal to 1160 (evaluated at 4 bar abs. and intermediate temperature of 350 °C). This value corresponds to Nu number 5.53 and heat transfer coefficient $357 \text{ W m}^{-2} \text{ K}^{-1}$ inside the coil, which is larger than h .

Fig. 3 shows the general trend at increasing the fluidization velocity, namely an initial rapid rise of h followed by a rather constant value. This is the consequence of the transition from an incipiently fluidized bed to a vigorously bubbling, well mixed bed. In contrast, the heat transfer coefficient is very sensitive to the particle size, fine bed largely improving h . The heat transfer coefficient attains a maximum value of around $280 \text{ W m}^{-2} \text{ K}^{-1}$. This value is in agreement with results reported by Di Natale et al. [14] and the theoretical estimates (see below). It is worth noting that the heat transfer coefficient in the fluidised bed results ten times higher than that in convective sections [15].

A combustion test of olive husks proved that no deposits were formed on the tube surface by fouling or condensation of compounds with low-melting temperature, the erosion exerted by the bed particles [16] preventing surface dirtying.

3. Integrated model of the fluidized bed and Stirling engine

For the sake of this preliminary activity, a fluidized bed combustor having a square geometry (300 mm size) is considered as combustion chamber for biomass fuels. The solid fuel, chips or pellets, is directly introduced into the fluidized bed at a mass flow rate F_f that depends on the assigned excess air ratio e . The fluidization velocity U is in excess with respect to the minimum fluidization velocity U_{mf} .

Fig. 4 shows the four basic blocks that were conceived for the implementation of the model: i) the fluidized bed heat transfer, ii) the fluidized bed combustion, iii) the heat recovery, iv) the Stirling engine conversion. Each block interacts with the others, the relevant variables of a single block being dependent on those of the other blocks as displayed in the diagram. Thus, a set of interconnected algebraic and differential equations results as

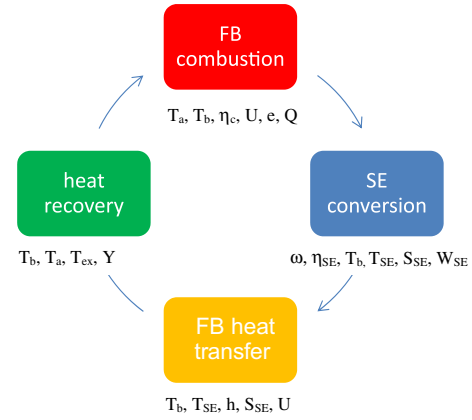


Fig. 4. Ideal blocks conceived for the mathematical model of the FB-SE integrated system.

mathematical model of the whole system. The analysis of each block is reported hereinafter. Although equations and correlations are taken from the literature as reported below, the model is innovative because it represents a first attempt in order to integrate characteristic equations of different components in a predictive and self-contained model.

3.1. Fluidized bed heat transfer

Following Kunii and Levenspiel [17], the general equation for computing the heat transfer coefficient h in a bubbling bed is given by Eq. (2).

$$h = \delta(h_r + h_g) + \left[\frac{1 - \delta}{\frac{1}{h_r + \frac{2k_{ew}}{d_p} + 0.05c_{pg}\rho_g U} + \frac{1}{h_p}} \right] \quad (2)$$

where h_r , h_g and h_p are the radiative, convective and single particle heat transfer coefficient, c_{pg} and ρ_g the specific heat and density of the gas, δ the bubble fraction, and k_{ew} the average thermal conductivity. The cited source [17] also provides the correlations for evaluating the unknown properties/parameters.

The power transferred from the bed to a generic immersed surface S is calculated via Eq. (3), as a function of the temperature difference $T - T_w$.

$$Q = hS(T - T_w) \quad (3)$$

The whole procedure for the calculation of the heat transfer coefficient has been implemented in a FORTRAN routine. Fig. 5 shows an example of the results in terms of h as a function of the bed particle size d_p , at three bed temperatures (700, 800 and 900 °C). The results of Fig. 5 indicate that the bed particle size exerts large influence on the heat transfer coefficient, small particles being preferable in order to improve the heat transfer rate. The bed temperature also affects h , which slightly increases with T_b .

3.2. Fluidized bed combustion

It is assumed that a biomass is fed to the combustor. Differently from solid fossil fuels, the content of volatile matters in biomass is high and the FB combustion can suffer of bypass and segregation phenomena, leading to depress the in-bed combustion efficiency

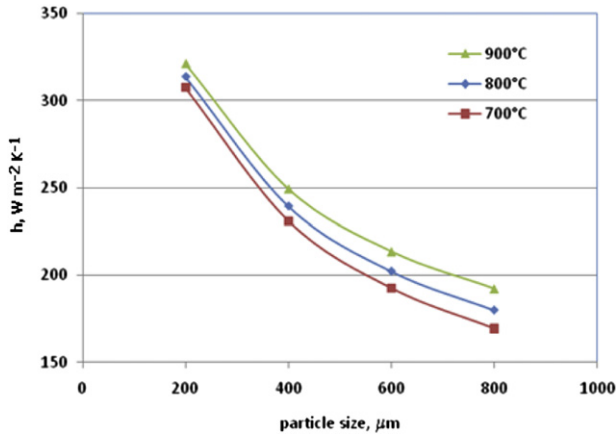


Fig. 5. Heat transfer coefficient in fluidized bed as function of the particle size and temperature (computed values).

[18]. Accordingly, an overall combustion efficiency η_c in the bed is introduced taking into account possible losses due to the fast release of volatile matters and imperfect mixing between combustible species [19]. The efficiency η_c is reasonably dependent on the bed temperature T_b , the fluidization velocity ratio U/U_{mf} and the excess air ratio e . Eq. (4) provides the relationship adopted by the model among the cited variables: an asymptotic dependence of η_c on T_b and e , accounting for the increased reaction rate, and a linear dependence on U/U_{mf} , accounting for better mixing. The biomass properties, namely nature, moisture, composition, particle size, are accounted for by the given parameters p_1 , p_2 and p_3 of Eq. (4).

$$\eta_c = 1 - p_1 \left(\frac{U}{U_{mf}} \right)^{-1} T_b \exp \left(-\frac{T_b}{p_2} \right) \exp \left(-\frac{e}{p_3} \right) \quad (4)$$

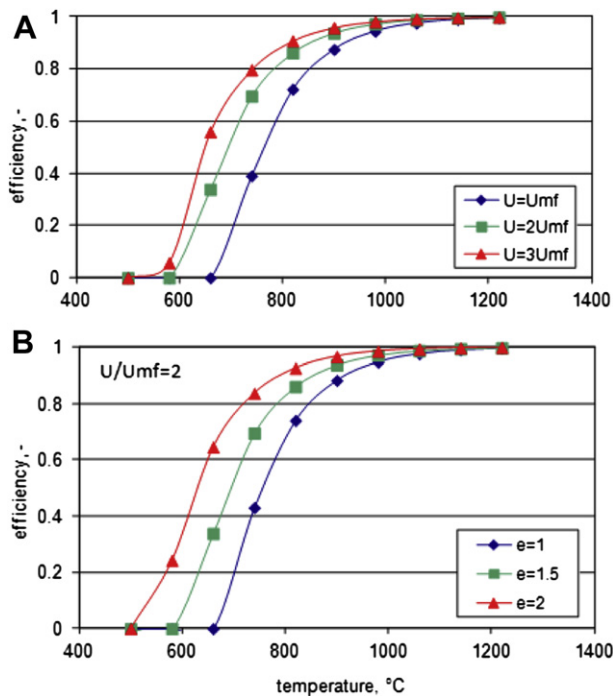


Fig. 6. Combustion efficiency in the bed versus FB temperature assuming the fluidization velocity (A) and excess air ratio (B) as parameter.

Fig. 6 displays the calculated curves of the efficiency versus the temperature for parametric values of the U/U_{mf} ratio (Fig. 6A) and the excess air ratio (Fig. 6B). The minimum fluidization velocity U_{mf} for particles of size d_p is computed via Wen and Yu correlation [20] reported in Eq. (5)

$$\frac{d_p U_{mf} \rho_g}{\mu} = \left[33.7^2 + 0.0408 \cdot \frac{d_p^3 \rho_g (\rho_s - \rho_g) g}{\mu^2} \right]^{0.5} - 33.7 \quad (5)$$

where g and μ are the acceleration of gravity and viscosity, respectively.

Adopting a lumped approach, the unsteady energy balance for the fluidized bed reads,

$$M_b c_{pb} \frac{dT_b}{dt} = AU \rho_a c_{pa} (T_a - T_b) + S_l K_t (T_0 - T_b) + h S_{SE} (T_{SE} - T_b) - F_f \eta_c \Delta H_c \quad (6)$$

where M_b is mass of the bed and A is the cross section. The terms at the right side, listed in the order as they appear, are: the enthalpy of the fluidizing gas stream $AU \rho_a$, the heat dispersed by conduction at the lateral wall S_l , the heat transferred to the Stirling engine throughout the surface S_{SE} , and the power of combustion $Q = F_f \eta_c \Delta H_c$.

3.3. Heat recovery

The heat recovery block consists of a heat exchanger downstream the combustion chamber, which makes possible the combustion air pre-heating by heat transfer with the hot flue gases. For instance, we can assume that finned tubes are transversally inserted in the zone of the column above the bed (freeboard), forming an air-to-air cross-flow exchanger. In this region the residual concentration of entrained bed solids would prevent the formation of deposits on the external surface of the exchanger [12].

The simplified equation of heat transfer (Eq. (7)) is derived assuming that the heat exchanger has very fast dynamics with respect to the fluidised bed, leading to absence of accumulation terms.

$$F_a c_{pa} (T_a - T_0) = F_g c_{pg} (T_b - T_{ex}) = Y \frac{T_b - T_a + T_{ex} - T_0}{2} \quad (7)$$

where T_0 and T_a are the temperatures of the air before and after heating, and T_{ex} is the temperature of flue gases after the exchanger. The average temperature difference $(T_b - T_a + T_{ex} - T_0)/2$ is the driving force, Y is a global coefficient of the heat exchanger, F_a and F_f the mass flow rates of the air and flue gas.

3.4. Stirling engine

Modular units of the Stirling engine are assumed to be coupled with the FB combustor. Each unit (1 kW nominal) is installed at the lateral surface of the fluidized bed. The geometrical properties, namely displacement and exposed surface of the tubular exchanger with the fluidized bed, are assigned. Helium is the working fluid at an average pressure of 8.8 MPa. The number of SE units N_{SE} that are connected to the fluidized bed can be assigned as model input parameter. The performance of the Stirling engine is predicted by using the software tool “StirlingPro.exe” developed by Normani [21]. The software accounts for different kinematic engine types and sizes, allowing for the calculation of the efficiency, power and torque for assigned geometrical parameters and physical properties

(e.g. pressure, upper temperature, lower temperature, fluid density/viscosity). The software tool has been iteratively executed in order to produce engine data-sheets as those reported in Fig. 7, showing the dependence of the mechanical power W_{SE} (A) and efficiency η_{SE} (B) on the angular speed ω for different upper temperatures. By interpolation of computed data, the 3rd order fitting equations (Eqs. (8) and (9)) are obtained.

$$W_{SE} = W_n [b_3^3(1 + n_3 T)\omega^3 + b_2^2(1 + n_2 T)\omega^2 + b_1(1 + n_1 T)\omega] \quad (8)$$

$$\eta_{SE} = 100 [q_3^3(1 + d_3 T)\omega^3 + q_2^2(1 + d_2 T)\omega^2 + q_1(1 + d_1 T)\omega + d_0 T] \quad (9)$$

It is also assumed that the dynamics of the Stirling engine is very fast with respect to that of the fluidized bed, thus its working conditions (η and W) can be computed on the basis of the current values of the bed.

The upper temperature T_{SE} is calculated via the differential energy balance (Eq. (10)) for the SE block:

$$C_{SE} \frac{dT_{SE}}{dt} = hS_{SE}(T_{SE} - T_b) - \frac{W_{SE}}{\eta_{SE}} \quad (10)$$

where C_{SE} is the heat capacity of the engine and S_{SE} the exposed surface.

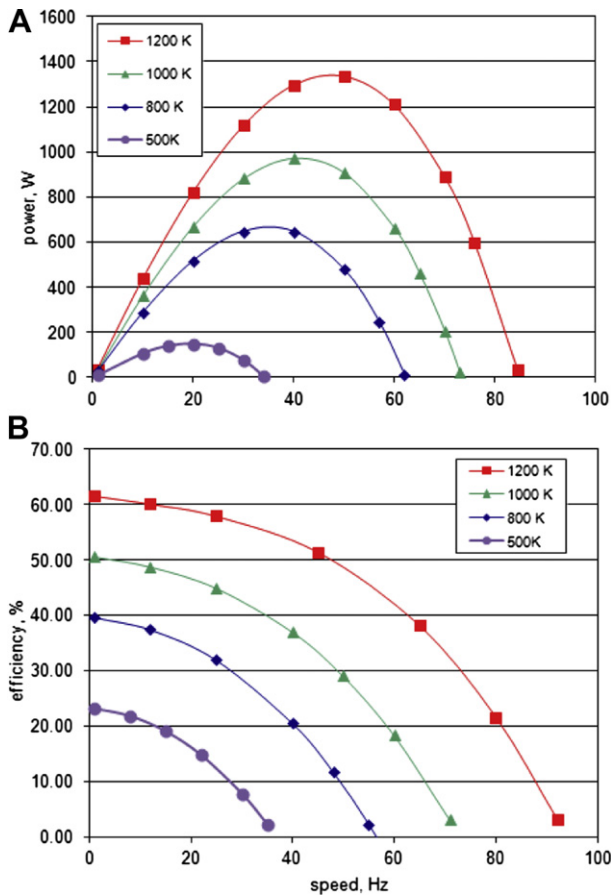


Fig. 7. Mechanical power (A) and efficiency (B) of a Stirling engine as function of the angular speed and upper temperature.

Table 1

Input variables and parameters of the model (base case).

Combustor size	m	0.3
Bed height	m	0.4
Bed particle size	m	0.3
Bed voidage (at minimum fluidization)	—	0.5
Particle density	kg/m ³	2600
Particle specific heat	J/kg/K	800
Air flow rate	kg/h	50
Fuel heating value	MJ/kg	17,000
Air for combustion	kg/kg	6.24
Fuel parameter p_1	K ⁻¹	20
Fuel parameter p_2	K	90
Fuel parameter p_3	—	0.8
Coefficient of the heat exchanger	W/K	7.5
Heat transfer coefficient (gas)	W/m ² /K	40
Number of SE modules	—	4
SE displacement	L	0.388
SE surface (1 module)	m ²	0.18
Heat capacity of SE	J/K	5000
Proportional constant	kg/s ² /K	−0.1
Integrative constant	kg/s/K	−0.0002
Derivative constant	kg/s ³ /K	5.0

3.5. Automated control

A control block is taken into account for assuring stable working conditions or improving the dynamic response of the system. It is assumed that in presence of a temperature gap, the fuel flow rate is adjusted by acting on the final control element (e.g. the frequency of a rotating screw feeder). The controller strategy is single-loop feedback PID [22], where the controlled variable is the bed temperature and the manipulated variable is the fuel flow rate. The controlling equation (Eq. (11)) links together the bed temperature T_b , the set-point temperature T_{set} and the fuel flow rate F_f .

$$\frac{dF_f}{dt} = G_1(T_{set} - T_b) + G_2 \frac{d(T_{set} - T_b)}{dt} + G_3 \int_0^t (T_{set} - T_b) dt \quad (11)$$

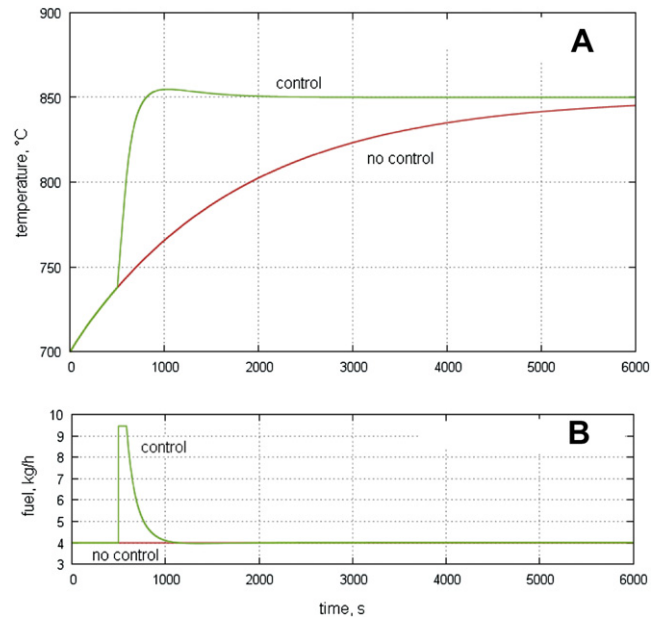


Fig. 8. Time profiles of the bed temperature (A) and fuel flow rate (B) without any Stirling engine connected to the fluidized bed: comparison between control and no-control operation.

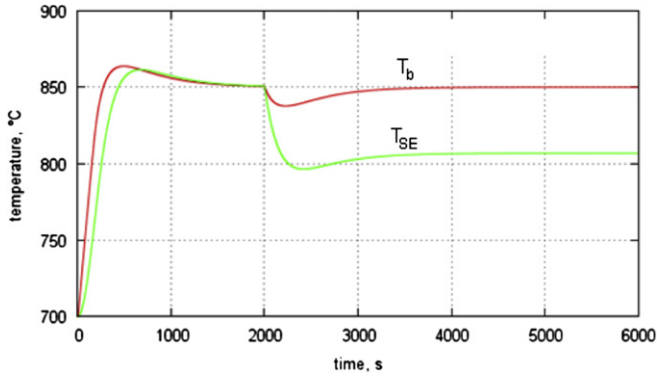


Fig. 9. Time profiles of the bed and SE temperatures under control operation ($N_{SE} = 4$, time of engine start = 2000 s).

Furthermore, the control action is limited by the following two constrains:

- The intervention of the integral control occurs within a narrow band close to the set-point;
- The lower limit of the excess air ratio is equal to 1.0.

3.6. Computation technique

The equation system includes three differential equations (Eqs. (6), (10) and (11)) that are numerically resolved over the time by adopting an explicit Eulerian scheme of integration. The set of initial conditions includes the temperature of the fluidized bed ($T_{b,0}$), the temperature of the Stirling engine ($T_{SE,0}$) and the fuel flow rate ($F_{f,0}$), for assigned air flow rate. The properties of gas, bed materials and fuels were taken from the literature [15,17,19]. Standard values of the operating variables and parameters are reported in Table 1. The values of the parameters for the combustion efficiency (p_1, p_2, p_3) were selected with reference to a high-volatile, high-reactive fuel, like a ligneous biomass.

3.7. Results

Fig. 8 displays the time dependent profiles of the bed temperature (panel A) and the fuel feeding rate (panel B) as evaluated by the model. The initial temperature is 700 °C and no Stirling engines ($N_{SE} = 0$) are connected to the combustor. Two curves are plotted for both cases of free (i) and controlled (ii) system. In absence of control, the temperature slowly rises up, achieving the thermal equilibrium value of around 850 °C. In presence of control, which is

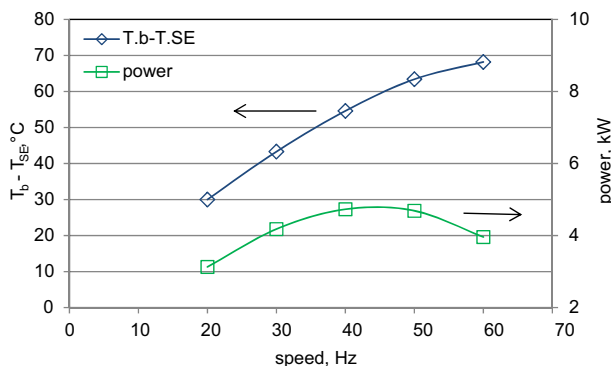


Fig. 10. Temperature difference between bed and Stirling engine and mechanical power versus the engine speed (control operation).

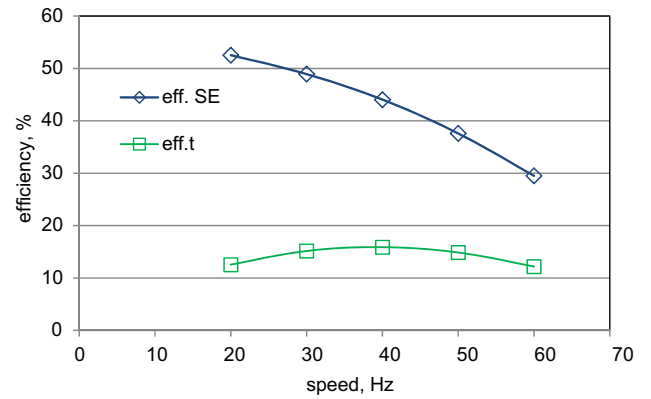


Fig. 11. SE efficiency and total efficiency versus the engine speed (control operation).

enabled at time $t = 500$ s, T_b curve rapidly increases, passing through a single overshoot, and promptly approaches the set-point value 850 °C. The control action clearly appears in Fig. 8 (panel B) with the sudden jump of the fuel feeding rate at time 500 s.

Fig. 9 displays the time dependent profiles of bed and Stirling engine temperatures for a simulation in presence of control. The Stirling engines ($N_{SE} = 4$) are started at time 2000 s, as indicated by the sudden decrease in both temperatures. Before this time the angular speed of the engines is set to zero; consequently, no heat is transferred from the fluidized bed to the engine via thermodynamic cycle. Only a limited heat transfer occurs for heating up the mass of the Stirling engines, as denoted by the shift of T_{SE} curve with respect to T_b curve. After $t = 2000$ s, the control action drives the system toward a steady working regime that is achieved upon $t = 4000$ s. In that case, T_{SE} is substantially lower than the bed temperature, due to the large thermal power required by the working Stirling engines. It is worth noting that the engines act as heat removal devices for the fluidized bed, similarly to tubes for steam generation.

More specifically, Fig. 10 shows the temperature difference between the bed and the Stirling engine along with the mechanical power as function of the engine speed for steady working regime. Similarly, the efficiency of the Stirling engine (η_{SE}) and the total mechanical efficiency (η_t) are displayed in Fig. 11. The larger is the engine speed, the higher the temperature difference, the lower the SE efficiency. This is consequence of the increased power to be transferred from the fluidized bed and of the intrinsic mechanical characteristics of the Stirling engine, reported in Fig. 7. For the simulated system a maximum of the mechanical power occurs at $\omega = 45$ Hz, whereas the maximum of the total efficiency is located at $\omega = 40$ Hz.

The time required for achieving steady working regime after engine start-up is plotted in Fig. 12 versus the number of SE modules and the engine speed. The condition $T_b - T_{set} = \pm 2$ °C is

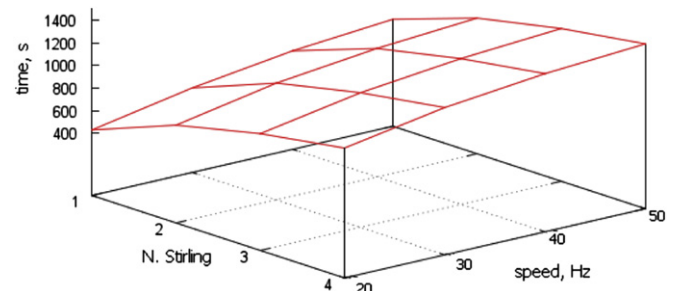


Fig. 12. Time required for achieving steady working regime after engine start-up versus SE number and engine speed (control operation).

assumed for setting steady working conditions. A rather long time in the order of 1000 s is required for achieving steady conditions, the thermal inertia of the fluidized bed being the limiting factor for rapid changes. It also appears that the system dynamics is much slower when the number of Stirling engines or the angular speed increase, corresponding to conditions of higher thermal power to be transferred from the fluidized bed to the engines.

4. Conclusions

Experiments were carried out for measuring the heat transfer coefficient in fluidized bed under conditions similar to those of FB-SE integrated system. The heat transfer coefficient attains a maximum value of around $280 \text{ W m}^{-2} \text{ K}^{-1}$. This value is in agreement with the theoretical estimates and results much more high than heat transfer coefficients in convective exchangers, in the typical range $20\text{--}50 \text{ W m}^{-2} \text{ K}^{-1}$. The obtained values can be reasonably extrapolated to a real FB-SE system. No deposits were formed on the tube surface by fouling or condensation after the combustion of a biomass, proving the reliability of the proposed solution.

An integrated model of fluidized bed combustion coupled with a Stirling engine was developed. The model provides as relevant outputs the bed temperature, the mechanical power and the efficiency of the Stirling engine, at changing the operating conditions and geometrical parameters.

Realistic values of the mechanical power generated by the engine are in the range $1.0\text{--}5.0 \text{ kW}$ for a $20 \text{ kW}_{\text{th}}$ fluidized bed combustor. These results indicate that the proposed “integrated system” is of interest for micro-scale co-generation from renewable fuels in connection with smart grids. However the slow dynamics of the system addresses toward applications characterized by rather steady request of mechanical/electrical power.

The automated control at changing the operation mode of the system was taken into account by choosing as manipulated variable the fuel flow rate. On the basis of the adopted variables and parameters, a prompt response is predicted when the load of the system is suddenly changed.

The results reported in this paper open perspectives for the development and operation of an experimental prototype for micro-generation to be fed with biomass fuels. This option is greatly appealing for appliances in rural and developing areas. In this concern, an experimental prototype of the integrated system is under construction at CNR for proving the proposed option and validating the model.

Nomenclature

A	section, m^2
b_i	parameters for SE power, s
c_p	specific heat, $\text{J kg}^{-1} \text{ K}^{-1}$
C	heat capacity, J K^{-1}
d_i	parameters for SE efficiency, K^{-1}
d_p	particle size, m
D_t	tube diameter, m
e	excess air ratio, —
F	mass flow rate, kg s^{-1}
g	gravity acceleration, m s^{-2}
G_1	proportional control constant, $\text{kg s}^{-2} \text{ K}^{-1}$
G_2	derivative control constant, $\text{kg s}^{-1} \text{ K}^{-1}$
G_1	integral control constant, $\text{kg s}^{-3} \text{ K}^{-1}$
h	heat transfer coefficient, $\text{W m}^{-2} \text{ K}^{-1}$
k_{ew}	average thermal conductivity, $\text{W m}^{-1} \text{ K}^{-1}$
K_t	conduction coefficient of the insulation layer, $\text{W m}^{-2} \text{ K}^{-1}$
L_t	tube length, m
M	mass, kg

n_i	parameters for SE power, K^{-1}
p_1	parameter of combustion efficiency, K^{-1}
p_2	parameter of combustion efficiency, K
p_3	parameter of combustion efficiency, —
q_i	parameters for SE efficiency, s
P	pressure, bar
Q	thermal power, W
S	surface, m^2
t	time, s
T	temperature, K
U	fluidization velocity, m s^{-1}
U_{mf}	minimum fluidization velocity, m s^{-1}
Y	global coefficient of the heat exchanger, W K^{-1}
W	mechanical power, W

Greek symbols

δ	bubble fraction, —
ΔH_c	combustion enthalpy, J kg^{-1}
η_c	combustion efficiency, —
η_{SE}	mechanical efficiency of the engine, %
η_t	total efficiency, %
θ	time, s
μ	viscosity, Pa s
ρ	density, kg s^{-1}
ω	angular speed, s^{-1}

Subscripts

a	air
b	bed
ex	exchanger
f	fuel
g	gas
l	lateral side
n	nominal value
p	particle
r	radiative
set	set point
w	wall
0	initial value

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